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Nelium, a refrigerant with high potential for the temperature range between 27 and 70 K

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Abstract

In the search for the optimum process for the liquefaction of hydrogen, it was found that mixtures of helium and neon, called "Nelium", allow processes with very high efficiency compared to pure helium or pure neon. This is demonstrated in the design of a 500 kW refrigerator between 40 and 60 K, whereby the composition is varied between pure helium and pure neon. It turns out that helium-rich mixtures have an advantage for the heat exchange, whereas the neon-rich mixtures are easier to compress in turbo compressors. In any case a process efficiency of over 44% is feasible.

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Keywords: hydrogen liquefaction; HTSC cables; radiation screen cooling; neon; nelium; turbo compressor

Nomenclature

A	cross section of heat exchanger passage	n	molar flow rate; rotational speed
D	diameter of turbine or compressor wheel	n_s	specific speed of turbine or compressor
NPH	number of pressure heads	Δp	pressure drop in heat exchangers
NTU	number of transfer units	u	circumferential speed
P; Q	refrigerator input power; refrigeration rate	x	neon content in mixture
Q_3	turbine outlet volumetric flow rate	η	cycle thermodynamic efficiency
f; j	friction factor; Colburn factor	η_s	turbine or compressor isentropic efficiency
g	geometry parameter heat transfer	η_f	fin efficiency
Δh_s	isentropic enthalpy difference	η_{distr}	pressure drop in distributors
m	mass flow rate	ψ	compressor flow parameter

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1. Introduction

In the past there was little need for large-scale refrigeration in the temperature range between 30 and 70 K. The situation has changed recently with the need for refrigeration for the following applications:

- Large-scale liquefaction of hydrogen,
- Electric power cables based on HTSC material,
- Beam screen as heat dump for synchrotron radiation in large proton accelerators [1].

For these three applications a refrigeration rate in the order of 500 kW is required. If one assumes a middle refrigeration temperature of 50 K, an ambient temperature of 300 K and a Carnot efficiency η, the drive power of such a refrigerator would be in function of the efficiency of the plant

$$P = \frac{1}{\eta} * \frac{T_a - T_0}{T_0} * Q = \frac{1}{\eta} * \frac{300-50}{50} * 500\ kW = \frac{2500}{\eta}\ kW. \quad (1)$$

According to Strobridge [2] the cost of cryogenic refrigerators is proportional to the 0.8 exponent of the power requirement. Fig 1 shows the required power and cost of such a plant in function of the efficiency. The cost is presented relative to the cost of a plant with an efficiency of 0.33.

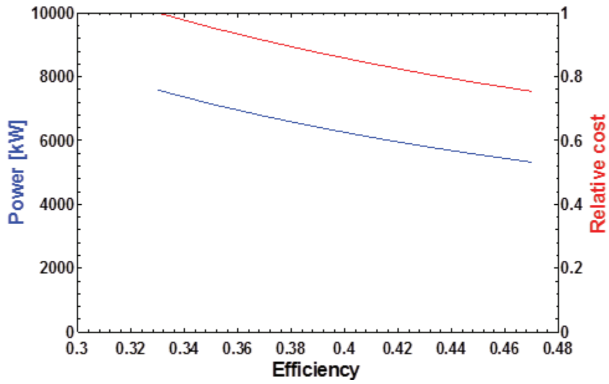


Fig. 1. Power consumption and relative cost of a 500 kW at 50 K refrigerator in function of the plant efficiency.

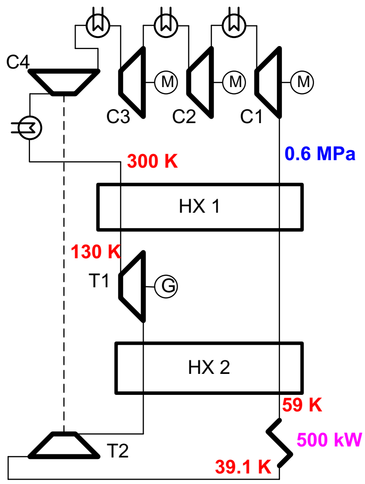
It is obvious that an increase in efficiency has a large positive influence, both on the investment and operating cost. Therefore it is worthwhile to investigate, how the efficiency can be increased above the value for present day technology. The efficiency depends on one hand on the choice of the cycle and the refrigerant and on the other hand on the available components.

2. Choice of temperature range, capacity, cycle, boundary conditions and component efficiencies

As an example we have chosen a 500 kW refrigerator, which cools a load from 60 to 40 K. For this temperature range there is no real alternative to a Brayton cycle with compression at ambient temperature, counter-current heat exchange and the production of refrigeration by work extracting expansion. Fig. 2 shows the chosen process, boundary conditions and component efficiencies, which have been used for the initial process calculations.

The refrigerant is compressed in a three-stage turbo-compressor with inter- and after-coolers, followed by a booster compressor C4, which is driven by turbine T2. The coldbox contains two cryogenic heat exchangers HX1 and HX2. The turbine T1 is introduced to cover the losses of exchanger HX1, whereas turbine T2 produces the refrigeration. The power of T1 is about five to ten times smaller than the power of T2. Therefore no direct power recovery is foreseen. The flow diagram shows a generator brake, but this power recovery is not considered in the overall energy balance, because it is only about 1-2 % of the main compressor input power.

The suction pressure of the main compressor has a large influence on the compactness of the plant, i.e. the volume of the heat exchangers and the diameter of the pipes, as well as on the efficient operation at part load [3]. For this example a pressure of 0.6 MPa has been chosen.



Refrigeration rate: 500 kW
Temperatures:
Outlet of all water coolers: 300 K
Inlet of turbine T1: 130 K
Outlet of turbine T2: 39.1 K
Outlet of load: 59 K
Pressure level:
Inlet to main compressor: 0.6 MPa
Isentropic efficiencies:
All compressor stages: 80 %
Turbine T1: 80 %
Turbine T2: 85 %
Parameters dependent on gas composition:
Neon content: x [%]
Pressure drop in all heat exchangers:
$\Delta p = 0.01 + 0.0001*x$ MPa
NTU of both cryogenic heat exchangers:
NTU= 40 – 0.1*x

Fig. 2. Choice of cycle, boundary conditions and component efficiencies.

For the first iteration the efficiencies of turbines and compressors have been chosen to be independent on the choice of refrigerant. Probably these values have to be modified later, when more details of these machines are available. For the heat exchangers a dependence of the NTU and the pressure drop on molecular weight of the refrigerant has been assumed. The reason for this will be explained in the heat exchanger chapter.

3. Choice of refrigerant

For this temperature range one has the choice between helium, hydrogen, neon and mixtures of these gases. Hydrogen is an option for large hydrogen liquefiers, but does not offer any advantages for other applications. So this leaves the choice between helium, neon and their mixtures [4]. Table 1 shows a number of their properties:

Table 1. Properties of helium and neon.

		Helium	Neon	Factor
Molar mass	kg/kmol	4.0026	20.18	5.04
Critical point	MPa/K	0.22/5.2	2.654 / 44.4	
Boiling point	MPa/K	0.1/4.2	0.1/ 27.15	
Triple point	MPa/K		0.043 / 24.56	
Density at 0.1013 MPa, 273.15 K	kg/m^3	0.178	0.900	5.04
Thermal conductivity	W/Km	0.146	0.045	0.31
Viscosity	10^{-3} Pa s	0.019	0.029	1.57
Specific heat c_p	kJ/kg K	5.193	1.030	0.20
Pr		0.664	0.667	1.00
Cost	EUR/Nm^3	10	200	20
World production	10^6 Nm^3/a	150	0.6	0.004

Neon is more expensive than helium; but the refrigerant inventory of such a plant is only about 2000 Nm^3. Near ambient temperature both helium and neon are nearly ideal gases with a molar specific heat of 20.8 kJ/kmol K. But the lower the temperature gets, the more real gas properties become pronounced, especially for neon. Fig. 3 shows the molar specific heat of the considered refrigerants at 0.6 MPa between 40 and 60 K.

There is no simple relationship for the transport properties for the gas mixtures. The Prandtl number has the ideal gas value of 0.667 for pure helium and pure neon. But according to [5] there is a minimum in the Prandtl number of the mixtures around a neon content of 25%.

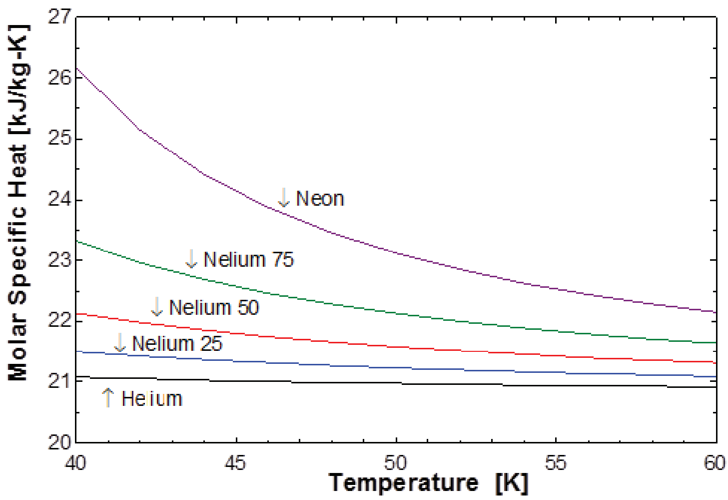


Fig. 3. Molar specific heat c_p at 0.6 MPa in function of temperature and gas composition.

4. Results of process calculations

If helium and neon were ideal gases with constant specific heat and one would assume the same NTU of the heat exchangers, the same isentropic efficiency of compressors and turbines and the same pressure drop in the heat exchangers, then the processes would be totally identical. But in reality there are differences as shown for the specific heat in Fig.3. Since the molar specific heat of neon is larger than that of helium, the needed molar flow rate to cover the refrigeration rate of 500 kW is different. Other differences come from the lower NTU and higher pressure drop needed for the neon-rich heat exchangers.

Table 2. Results of process calculations.

		Helium	Nelium 25	Nelium 50	Nelium 75	Neon
Mol. Weight	kg/kmol	4.0026	8.047	12.091	16.135	20.179
Mass flow	kg/s	4.790	9.485	13.964	18.022	21.138
Molar flow	kmol/s	1.197	1.179	1.155	1.117	1.048
NTU_{HX}		40	37.5	35	32.5	30
Δp_{HX}	MPa	0.01	0.0125	0.015	0.0175	0.02
p_{in}	MPa	0.6	0.6	0.6	0.6	0.6
p_{out}	MPa	2.345	2.430	2.520	2.665	2.970
Compressor Power	kW	5678	5772	5834	5897	5984
Exergy	kW	2650	2662	2672	2684	2699
Efficiency		0.467	0.461	0.458	0.455	0.451

The differences in NTU and allowed pressure drop in the heat exchangers were already mentioned in Fig.2 and will be explained in the heat exchanger chapter. The two pressures p_{in} and p_{out} refer to the main compressor. The compressor power is the sum of the power of the three sections of the main compressor shown in Fig.2. The exergy of refrigeration is somewhat larger for the neon-rich refrigerants because of the not constant specific heat values shown in Fig.3. The efficiency is calculated by dividing the exergy by the compressor power.

Table 2 shows, that there is a slight reduction in efficiency with the increase in neon content. This is due to the larger losses in the heat exchangers. This picture may change, when differences in the isentropic efficiency of turbines and compressors are taken into account.

5. Cryogenic heat exchangers

According to Ruehlich [6] there is a simple relationship between the heat transfer measured in NTU of a heat exchanger stream and the pressure drop measured in number of pressure heads NPH:

$$\frac{NPH}{NTU} = \frac{f*\frac{L}{D_h}}{\frac{Nu}{RePr}*\frac{4L}{D_h}} = \frac{f*Re}{4\,Nu}*Pr = \frac{f}{j}*\frac{Pr^{\frac{2}{3}}}{4} = g*\left(\frac{Pr}{0.7}\right)^{\frac{2}{3}} \quad (2) \qquad \Delta p = NPH*\frac{\varrho}{2}w^2 = g*\left(\frac{Pr}{0.7}\right)^{\frac{2}{3}} NTU*\frac{m^2}{2\varrho A^2}. \quad (3)$$

Ruehlich has evaluated the parameter g for several different heat exchanger geometries. For cryogenic heat exchangers the geometry "Serrated Plate-Fin" with g = 3 is typical. Some of the available pressure drop has to be reserved for the internal distributors and the headers. For this a distribution efficiency of η_{distr} with a value of about 0.8 is introduced. So the cross flow area needed for one of the streams in a heat exchanger is

$$A_i = m*\sqrt{\frac{g*\left(\frac{Pr}{0.7}\right)^{2/3} NTU_i}{2\varrho\Delta p_{all}}} = n*\sqrt{\frac{g*\left(\frac{Pr}{0.7}\right)^{2/3} NTU_i*\Re*T*M}{2*p*\eta_{Distr}*\Delta p_{all}}}. \quad (4)$$

Normally the NTU of the total heat exchanger is given. For a two-stream counter-current exchanger, the NTU value of a single stream is two times the overall NTU divided by the fin efficiency in this passage:

$$NTU_i = \frac{2}{\eta_f} NTU_{total} \quad (5) \qquad NTU_{LP} = \frac{2}{0.9}*40 = 88.9. \quad (6)$$

For the LP channel of the plate fin heat exchanger HX1 of Fig.2 the following flow area results for the different refrigerants:

Table 3. Cross section of the low pressure channel of exchanger HX1 in Fig.2 for the different refrigerants.

Refrigerant		Helium	Nelium 25	Nelium 50	Nelium 75	Neon
M	kg/kmol	4.0026	8.047	12.091	16.135	20.179
n	kmol/s	1.197	1.179	1.155	1.117	1.048
Pr		0.667	0.570	0.590	0.630	0.667
NTU_{HX}		40	37.5	35	32.5	30
NTU_{LP}		88.9	83.3	77.8	72.2	66.7
Δp_{HX}	MPa	0.01	0.0125	0.015	0.0175	0.02
A_{LP}	m^2	0.620	0.712	0.763	0.777	0.746

General: T = 300 K, g = 3 (serrated fins), $\eta_f = 0.9$, $\eta_{distr} = 0.8$, R=8.314 kJ/kmol K

The cross section of the full exchanger is about two times the value of the free-flow area of the low pressure stream. Even though a lower NTU value and a higher pressure drop were allowed for the neon-rich streams, the cross section of the heat exchanger increases. This shows that from the heat exchanger side pure helium is the best refrigerant.

6. Turbine

Here we want to establish a few geometrical parameters of the turbine T2 in flow diagram Fig.2. The optimum circumferential speed of a radial turbine is given by

$$u_{2,opt} = 0.7*\sqrt{2*\Delta h_s} \quad (7) \qquad u_{opt}\left(\frac{m}{s}\right) = 0.7*\sqrt{2000*\Delta h_s\left(\frac{kJ}{kg}\right)}. \quad (8)$$

Expansion turbines are characterized by a parameter called specific speed n_s. It is defined by the following formula:

$$n_s = C*\frac{n*\sqrt{Q_3}}{\Delta h_s^{3/4}} \quad (9) \qquad n_{opt} = \frac{n_{s,opt}*\Delta h_s^{3/4}}{C*\sqrt{Q_3}} \quad (10) \qquad D_{opt}(m) = \frac{u_{opt}\left(\frac{m}{s}\right)}{60*\pi*n(rpm)}. \quad (11)$$

When one has established the optimum circumferential speed from (8) and the optimum rpm from (10), one can find the optimum diameter of the wheel with (11).

Table 4. Parameters of the turbines T2 belonging to the different refrigerants.

Turbine		Helium	Nelium 25	Nelium 50	Nelium 75	Neon
Power	kW	560.9	540.9	520.2	497.0	468.9
Δh_s	kJ/kg	137.8	67.1	43.8	32.4	26.1
V^3	m^3/s	0.626	0.595	0.560	0.521	0.461
u	m/s	367.4	256.4	207.2	178.3	159.9
D	(m)	0.15	0.175	0.19	0.2	0.2
n	rpm	46808	27998	20842	17036	15279
Brake compressor						
Δh_s	kJ/kg	91.41	45.14	29.39	21.97	17.39
V_0	m^3/s	1.287	1.224	1.16	1.06	0.892
D	m	0.18	0.205	0.22	0.235	0.235

The revolution speed n of the brake compressor is dictated by the turbine. The diameter of the brake compressor wheel has been chosen so that

$$\psi = \frac{u}{\sqrt{2 * \Delta h_s}} \cong 1{,}0. \tag{12}$$

From Table 4 one recognizes that the turbines in the neon-rich cycles have larger diameters than the helium turbine and they run much slower. Probably the efficiency of the turbine will increase with the neon content in the refrigerant. But this is proprietary know-how of turbine manufacturers.

7. Main compressor

For the example shown in Fig.2 a main compression in three stages, i.e. with two inter- and one after-coolers has been chosen. The reason for the choice of three stages with intercooling is to come as close as reasonable to the isothermal compression. The good efficiency of turbo-compressors is the reason for the improvement in overall efficiency. But turbo-compressors for light gases like helium and neon require very high circumferential speed and multiple wheels. With a speed of 350 m/s one can obtain a pressure ratio of about 1.6 with neon and about 1.1 with helium per wheel. Table 5 shows some parameters of the first stage of compression, i.e. before the first intercooler.

Table 5. Parameters of the first stage of compression and subdivision into wheels.

Refrigerant		Helium	Nelium 25	Nelium 50	Nelium 75	Neon
m	kg/s	4.790	9.485	13.964	18.022	21.138
p_0	MPa	0.6	0.6	0.6	0.6	0.6
T_0	K	295.83	295.77	295.67	295.57	295.59
v_0	m3/kg	1.027	0.51	0.34	0.255	0.2036
V_0	m3/s	4.920	4.845	4.74	4.587	4.3030
p_3	MPa	0.955	0.9725	0.983	1.008	1.0400
Pressure ratio		1,592	1.621	1.638	1.679	1.733
Δh_s	kJ/kg	314.92	163.34	111.32	88.00	75.20
Subdivision into several wheels:						
Number of wheels		4	3	2	2	2
$\Delta h_{s,wheel}$	kJ/kg	78.7	54.4	55.7	44.0	37.6
$u_{2,opt}$	m/s	396.8	330.0	333.7	296.7	274.2
D_2 (m)	(m)	0.5	0.6	0.8	0.8	0.8
n	rpm	15165	10509	7969	7086	6550

For the pure helium compressor it is proposed to use four wheels in series for this first compression stage, whereas for pure neon two wheels should be sufficient. Still the speed of the helium compressor has to be more than two times higher than that of the neon compressor. Since the neon wheels are larger and the speed is slower, it can be expected that the efficiency of the compressors for the Neon-rich cycles is higher than that of the pure Helium cycle.

8. Summary

A Brayton cycle for a 500 kW refrigerator for the 40 to 60 K temperature range has been investigated with helium, neon and their mixtures as refrigerants. Helium has advantages for the heat exchangers, but neon-rich mixtures are easier to compress in turbo compressors. These turbo compressors are responsible for the increase of overall efficiency, which is predicted to be in the order of 45%.

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